An Alternate APD Cooling Proposal
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The current baseline plan for cooling the NOvA APD is to use Thermo-electric coolers which are in turn cooled by a small chilled water system. The chilled water system is in turn cooled by a larger chilled water system planned to be provided as part of the building. This larger chilled water system would be cooled by a water glycol loop and then a large refrigerant based chiller, both also provided as part of the building.

There are several problems with this scheme which have not been addressed or solved. These problems include:

1) Because it is currently planned to turn on one block of the detector approximately every one to three weeks, the larger chilled water system and the refrigerant system will need a turn down ratio of about 50:1. This simply CANNOT be met with conventional refrigerant equipment.

2) The hot side of the TEC cannot be cooled to a temperature of less than about 15 C because of the temperature differences needed across each heat exchanger and the requirement to keep the water above freezing temperatures. To keep the APD at -15 C, the TEC needs a temperature difference across of 30 C. At this temperature differential, approximately one to three TEC units are expected to fail per day[1].

3) One water-glycol and two chilled water loops all daisy chained together present a significant risk for a single point failure which will cripple the entire detector.

4) Based on experience with the many chilled water systems designed and built for the NuMI beamline, each chilled water loop will require approximately $50,000 of instrumentation, controls, and water treatment equipment to raise each system to an acceptable level of reliability.

5) Each heat exchanger will have a temperature approach across it, which is essentially an introduced inefficiency. Multiple heat exchangers in series waste energy.

To mitigate these fundamental design problems, I propose to directly cool the hot side of the TEC with refrigerant and to eliminate all of the chilled water systems.

Specific design decisions chosen to mitigate the above problems are:

1a). One commercially available refrigerant compressor and condenser will be used for each block. At 4 watts of heat load from each TEC and APD combination, this sums to about 1500 watts of refrigeration. Since the refrigeration industry typically uses U. S. Customary units, this equates to approximately 5000 BTU per hour. This is

[1] John Oliver, NOvA Note 35-v2, page 2

approximately the capacity of the smallest window air conditioners routinely sold to consumers, so OEM compressors, condensers and thermo-expansion valves (TXV) are widely available.

If this heat load is significantly different than 4 watts per APD, many different sizes of commercially produced refrigerant compressor and condenser units are available as catalog item to choose from. See: http://www.copeland-corp.com/cp_rf/prod_sol/cp_rf_products_condunits_SystemPro.htm for one OEM list of available hermetic condenser packages.

This choice solves the turn-down issue since as a block is completed, the refrigerator would be turned on and operated at the design load. Loads would not change as a function of time, because the next block to be completed would use a separate refrigerator.

2a). Freezing of the refrigerant liquid is not a concern at temperature under consideration. However, one does want to keep the low pressure side of the refrigerant system above atmospheric pressure. Different refrigerants have different bolting temperatures at atmospheric pressure. R134a boils at -15 F (-26 C). R-22 boils at -41 F (-40 C). So, by choosing the refrigerant and the low pressure correctly, one can tune the temperature of the hot side of the TEC. Presumably, one would choose the hot side temperature of the TEC to be cooler than 15 C so that the TEC is not forced to operate at a large temperature difference and the TEC lifetime would be extended.

3a). Since there would be fewer links in the cooling system chain and since the major equipment would be based on mass produced refrigeration components from reputable original equipment manufactures (OEM), the reliability would be improved. Because the APD and TEC units on a each block would be cooled by a single unit, the failure of any one unit would only affect one block of the detector. The other detector blocks would continue to operate normally.

4a). There would be fewer components in each refrigeration system. So, the amount of controls, instrumentation and fluid conditioning equipment would be substantially reduced. Water treatment for corrosion control would be eliminated (the commercial refrigerants are non-corrosive).

5a). Because the refrigerator would directly cool the items that need to be cooled, there would be no intermediate heat exchangers, each of which reduces the energy efficiency of the entire system.

Dozens of refrigerants exist for the temperature range of interest. However, one should limit the choices to those refrigerants which are widely used commercially and environmentally benign. Both R134a (the R-12 replacement widely used in automobiles) and R-22 (the refrigerant used in many refrigerators and air conditioning systems) have OEM equipment made specifically for them. Both are environmentally friendly. R134a has an ozone depletion potential (ODP) of 0.0 and R-22 has an ODP of 0.05. The now discontinued refrigerant R-12 (a.k.a. Freon) has an OPD of 1.0. R-22 production will be ceased in 2030. (Note this is ten years later than the date originally included in the Montreal protocol). New equipment will not be able to use R-22 after 2010. Because the NOvA schedule is on par with the R-22 new equipment phase-out, R134a is the preferred refrigerant choice.

R404 and R-410 (promoted by Carrier under the trade name Puron) are also environmentally friendly. The downside to these refrigerants is that the condensing pressure is significantly higher than R22 or R134a. In addition, there is less commercial equipment available intended specifically for these refrigerants. A good list of common refrigerants is available at:
http://www.refrigerants.dupont.com/Suva/en_US/pdf/k10909v2.pdf

The direct refrigerant cooling proposal would use the following equipment:

A) A refrigerant compressor and condenser assembly. This is similar to the unit located outdoors for most residential air conditioning systems but would be physically smaller (a typical home air conditioner provides 2 tons of cooling (24,000 BTU/hr) which is five times larger than the loads from the NOvA APD and TEC units.

B) A thermo-expansion valve (TXV) or a capillary tube. This separates the high pressure (hot) side (liquid side) of the cycle from the low pressure (cold) side of the system. Inexpensive refrigerators and air conditioners use capillary tubes. These have no moving parts, are very robust, but do not provide any load control. Thermo-expansion valves use a temperature sensing bulb to adjust the opening of a valve to increase or decrease the refrigerant flow. A TXV provides better temperature control but at the expense of a more complicated component. TXV units are commercially available for use with R-134a at evaporator temperatures of 25F, 20F, 10 F and -10F (-4C, -6.7C, -12C and -23C).

C) An evaporator. Typical refrigerant systems use a coil as an evaporator because they are attempting to remove heat from air. In the NOvA application, the evaporator would be quite small as it would be removing heat from a small copper block in contact with the hot side of the TEC.

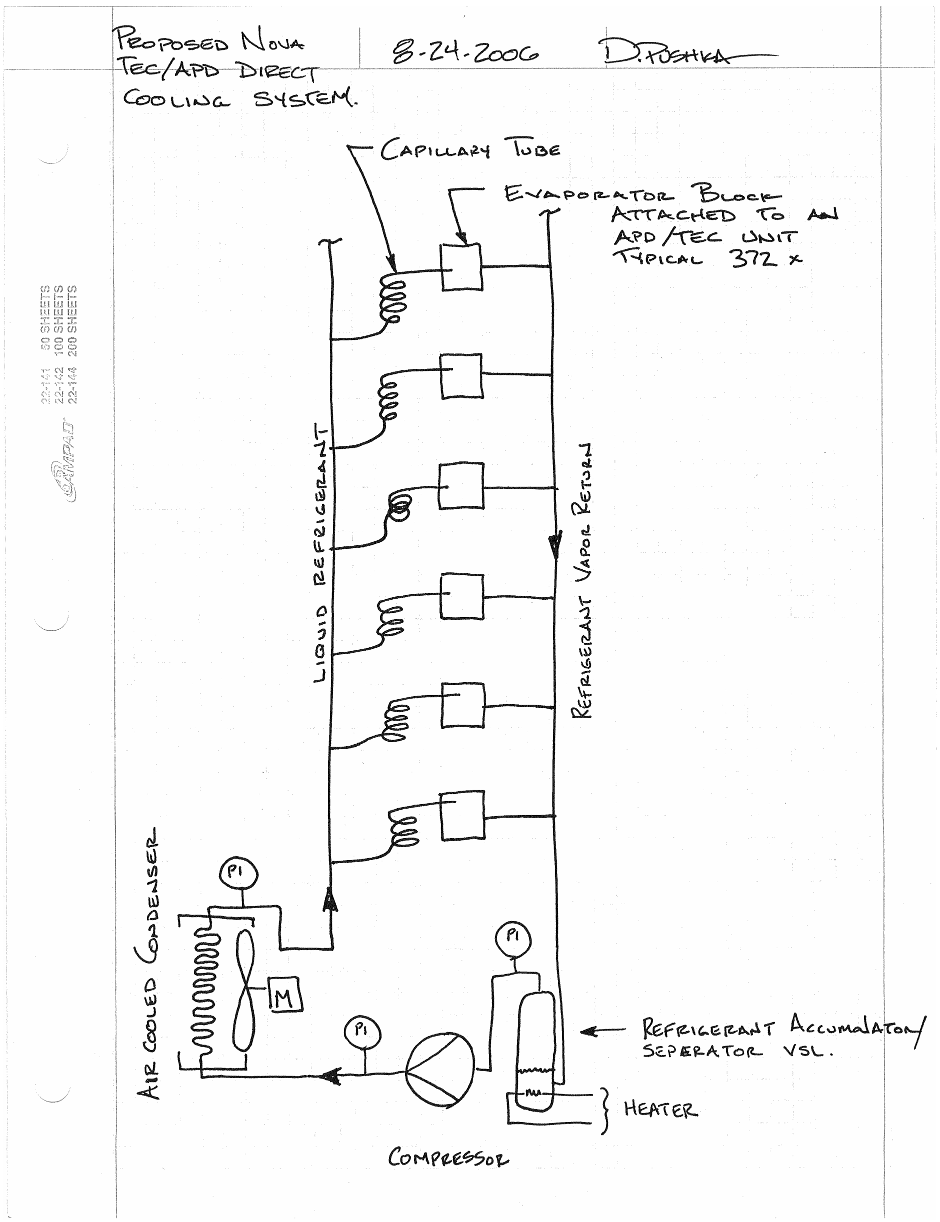
D) Instrumentation. A refrigerant filter-dryer unit, a refrigerant moisture indicator, and compressor suction and discharge gauges would be installed on each system to allow rapid evaluation of each refrigerator operating parameters.

Heat from a direct cooling system would be rejected in the condenser connected to the compressor discharge. The total energy to be removed from the condenser is the sum of the heat load from the TEC units, the energy absorbed in the refrigerant return piping, and the electrical energy used by the compressor. A 5400 BTU/hr compressor and condenser unit would have a ¾ HP motor on the compressor. So the total heat rejected by the unit would be about 7300 BTU/hour. The sum for 55 blocks worth of detector would be 402,063 BTU/hr. If rejected to the building air, this design would require about

33 tons of refrigeration – similar to the 30 ton DX unit shown in the building CDR drawing M-1. If the APD/TEC direct refrigerant cooler were to have the compressor and condenser units located outside, the heat would be directly rejected outdoors, eliminating the 30 ton DX unit cost.

It is also possible to purchase OEM compressor and condenser units designed to reject heat directly to water or to a water-glycol mix. Since water temperatures generally run cooler than the maximum air temperature, the water cooler units offer a higher capacity for the same size unit than do the air cooled units.

Direct Refrigerant Cooled APD-TEC schematic:



Each evaporator block is fed liquid refrigerant via a fine capillary tube from a liquid refrigerant supply header. On any given detector block, 372 evaporator blocks would be plumbed in parallel. Since the pressure drop in the liquid refrigerant supply header is very small compared to the pressure drop in the capillary, the variation in refrigerant flow from one evaporator block to another would be negligible. Because the refrigerant is warm liquid in the supply header, there is no concern about this line sweating or condensing water vapor from the room air.

On residential air conditioners, it is common to feed several parallel evaporator tubes with individual capillary tubes, although the number of parallel paths is usually less than ten.

Each evaporator block would be connected to a common refrigerant vapor return line. This refrigerant vapor is cold gas, perhaps with some liquid droplets. Unless insulated, this line will sweat until the refrigerant gas warms to above the dew point of the building air.

Until it is shown that the heat load from the APD and TEC units and the vapor return line is sufficient to vaporize all liquid refrigerant, it would be wise to plan to include a refrigerant accumulator / separator vessel before the compressor suction to be sure that only vapor enters the compressor suction. It is common to include refrigerant driers and moisture monitors in this low pressure portion of the system.

Future work:

The decision on where to locate the condenser unit needs to be made with due consideration for the impact on the building. Locating the unit outdoors reduces the building cooling heat load during the warmer months. Locating the unit indoors saves energy during the heating months because the waste heat from the APC and TEC direct cooling is captured in the building. Naively, because of the northern location, optimizing condenser location to capture the energy during winter heating months seems the most economical. But, input from the building folks is needed.

A small scale test of this cooling method should be conducted. Design and fabricate a few dozen copper evaporator blocks, attach resistance heaters to the evaporator blocks, braze copper capillary and suction line tubes, purchase a small compressor and condenser unit, evacuate and charge the unit and demonstrate that the copper evaporator blocks can be maintained at the desired temperature (perhaps -6 C) while the resisters apply a 4 W heat load to the blocks.